[논문] 태양에너지
Solar Energy
Vol. 10, No. 3, 1990

비등온 실린더 모델을 이용한 태양로의 강제 대류에 의한 열 손실 분석

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Forced Convection Modelling of a Solar Central Receiver using Nonisothermal Cylinders in Crossflow

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요 약

표면 온도가 균일하지 않은 원통을 균속도 유동장에 가로 놓았을 경우, 표면에서의 열전달 특성은 표면이 등온이거나 일정한 heat flux가 주어졌을 때와는 판이하게 다르다.

본 연구에서는 공기의 균속도 유동장내에서 두가지 경우(step형 및 선형변화)의 비등온 경계조건이 원통면을 따라 원주방향으로 주어졌을 때 표면에서의 열전달 특성을 고찰하였다. Step형 변화는 원통형 태양로의 표면에서 관찰될 수 있다. Solar One(California주의 Barstow시에 있는 태양로)의 경우, 작동유체(물)는 표면을 따라 원주방향과 수직으로 설치된 튜브를 따라 흐르면서 액체상태로부터 고온고압의 증기로 변한다. 이 과정에서 태양로 표면의 receiver panel은 그 위치에 따라, preheater, boiler, 그리고 superheater의 역할을 수행하며 표면의 온도도 균일하지 않은 분포를 나타낸다. 이와 같은 경우 표면의 평균 온도를 가지고 대류에 의한 열 손실을 계산하면 큰 오류를 범할 가능성이 있다.

ABSTRACT

When nonuniform thermal boundary conditions are imposed on the surface of a circular cylinder in crossflow, the heat transfer characteristics can be quite different compared to what is found for isothermal or constant heat flux boundary conditions. In

the present analysis, two kinds of nonuniform boundary conditions along the circumference of the cylinder are considered in a uniform stream of air: step changes and linear profiles. Step changes in temperature can arise on the surface of an external, cylindrical, solar central receiver. As the working fluid(water) flows through the vertical tubes that ring the circumference of Solar One(a solar central receiver in Barstow, California), the solar flux on the receiver heats the water from a liquid to a superheated state. In this process, portions of the receiver panels, and thus portions of the circumference of the cylinder, function as a preheater, boiler, or superheater. Hence the surface temperature can vary significantly around the cylinder. Common engineering practice has been to use an average wall temperature with an isothermal cylinder heat transfer coefficient when estimating the convective loss in these kinds of situations.

Nomenclature

Nu_{av} Nusselt number
Nu_{av} average Nusselt number
Re Reynolds number based on cylinder diameter and freestream velocity.
Tw wall temperature[K]
$T_{w,av}$ average wall temperature[K]
T bulk fluid temperature[K]
UWT uniform wall temperature boundary condition
UHF uniform heat flux boundary condition
angle measured from the front stagnation point of a cylinder[degree]
θ angle measured from the front stagnation point of a cylinder[degree]
Θ_w dimensionless wall temperature= $[T_w(\theta)-T_\infty]/[T_{w,av}-T_\infty]$

1. INTRODUCTION

To simulate the convective phenomena numerically can be quite difficult, particularly at Reynolds numbers over a few hundred, due to the complex transient flow behind the cylinder. Dennis and Chang[1] have studied the steady flow past a cylinder, in which they limited the range of investigated Reynolds numbers to 100. Borthwick has recently reported two studies of flow around a cylinder. In the first of these[2], a comparison was made of two methods of discretization of the governing equations. A range of Reynolds number up to 400 was reported, and particular attention was given to the numerical approach on the solution values.

Our desire here is to obtain solutions numerically over a wide range of Reynolds numbers for the nonisothermal conditions with rather simplified boundary conditions and to examine how closely they approximate the long term averaged heat transfer behavior in the real situation. This was done in an earlier paper[3] for both the uniform wall temperature(UWT) and the uniform heat flux(UHF) conditions. Excellent agreement was found between calculations and experimental valuse for wall shear and heat transfer, the latter being for both local and average values. Included were comparisons to the work or summaries of Eckert and Soehngen[4], Krall and Eckert[5], Sarma and Sukhatme[6], and Zukauskas and Ziugzda[7]. In the present analysis, we use this approach for predicting nonisothermal convection from circular cylinders.

2. METHOD OF ANALYSIS

The analysis is carried out as described in the earlier paper noted above. Surface temperature variations as a function of the angle from the front sagnation point were assumed as shown in Figure. 1. In addition, two cases were considered where a linear variation in surface temperature was assumed. One started at zero at the front stagnation point and increased, while a second started high and decreased to zero at the rear stagnation point. Note that all of these distributions are carefully chosen to yield the same average wall temperature, except for the UHF condition discussed in the Results section. Local Nusselt numbers were determined from numerically-computed temperature gradients at the wall for the range of Re between 200 and 3480 for each wall temperature distribution. This included the UWT and UHF cases described in the earlier paper. Local values of Nusselt number were then averaged according to the following:

$$Nu_{av} = [\int_0^{180} Nu(\theta) d\theta] / 180$$

3. RESULTS AND DISCUSSION

Results for a number of nonisothermal cases are given in Figures 2 and 3. As anticipated, cases which have higher temperatures over the front portion of the cylinder demonstrate higher average heat transfer.

In Fig. 2. each plot shows the angular distribution of the local Nusselt number for the corresponding boundary conditions in Fig. 1 at Re=200 and 1000. Although such conditions could be difficult to realize in practice, some detailed discussions are given here regarding discontinuities in heat transfer, as they display distinct characteristics with the location of step changes in wall temperature.

Fig. 2(a) shows, for Re=200, a small peak at $\theta = 90°$ and makes a sharp decrease to Nu= -52. The sudden small increase in heat transfer might be the effect of upstream diffusion, which subcools the oncoming fluid. This enhances the heat transfer. The negative Nusselt number

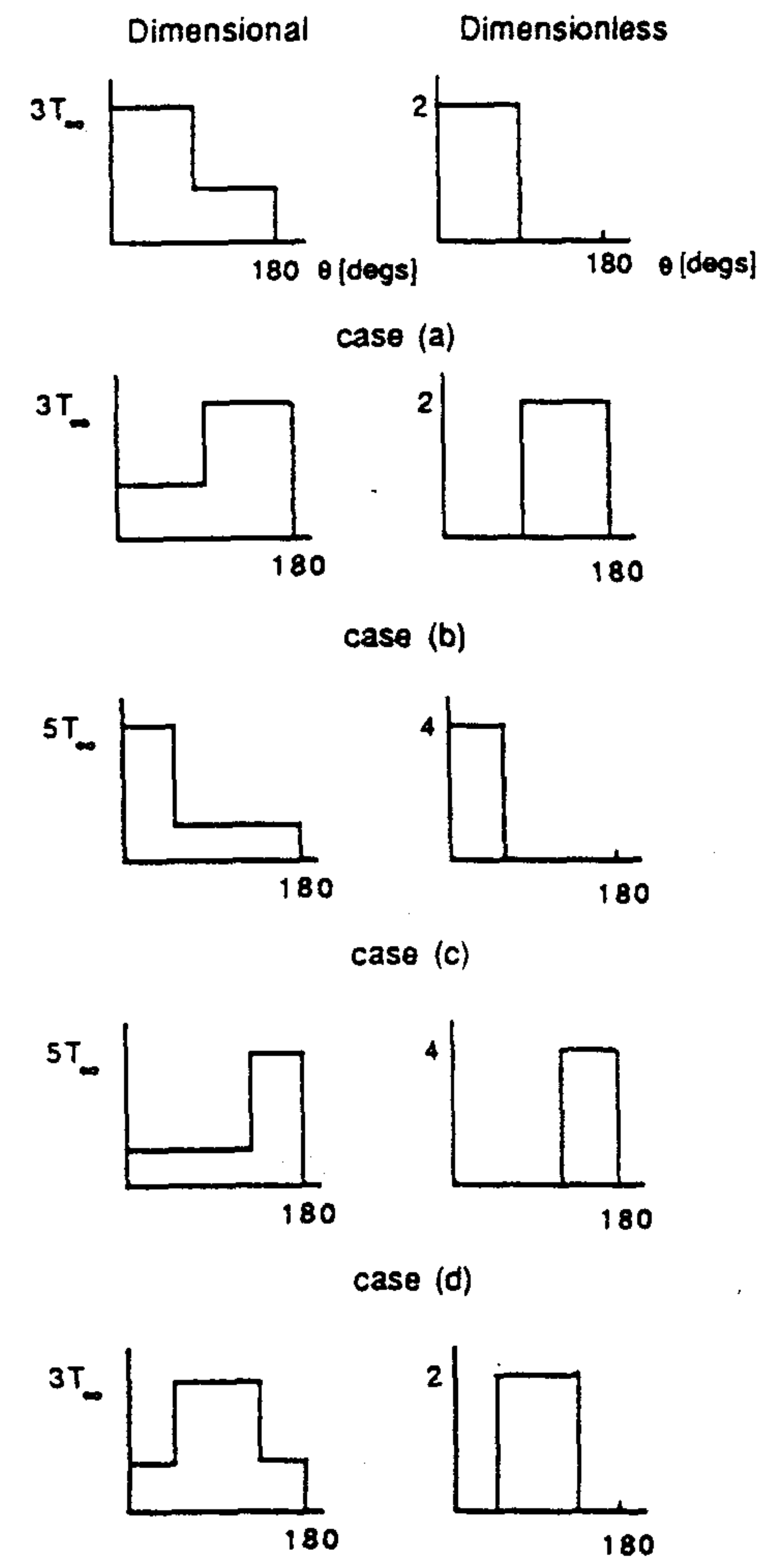
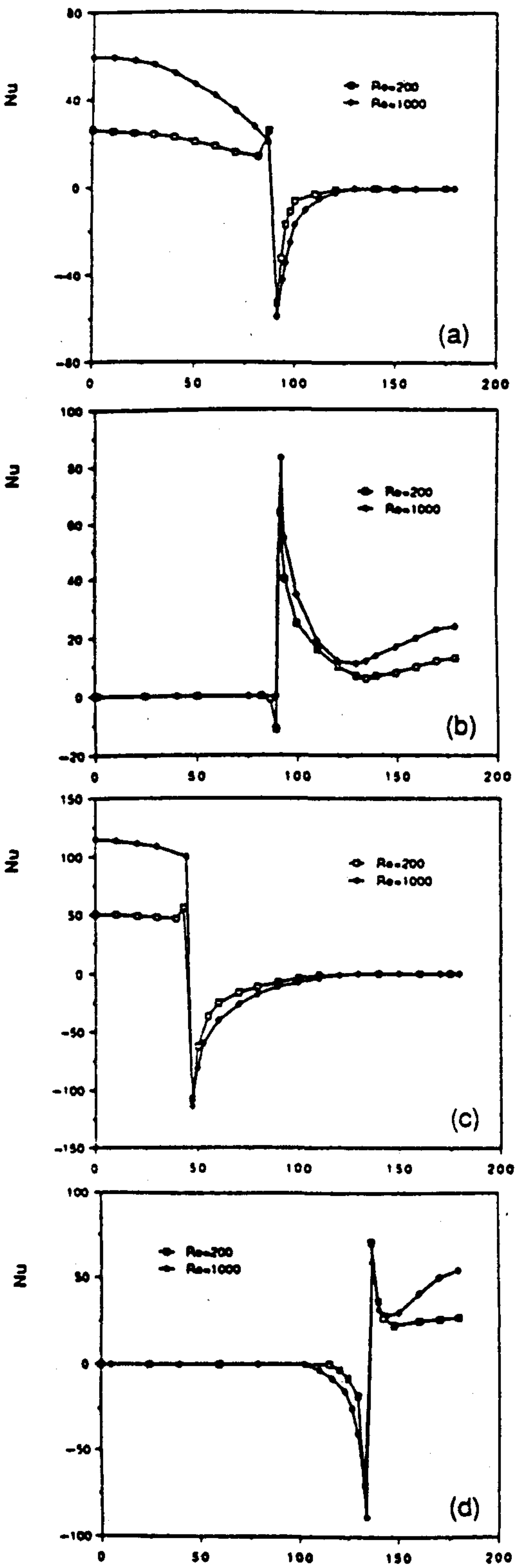


Fig.1 Stepchanges in wait temperature assumed in this anatysis; note that all cases have the same average temperature

observed here indicates a reversal in heat transfer direction. That is, heat flows from the fluid to surface. As the fluid flow proceeds further downstream toward the rear stagnation point, the Nusselt number asymptotically approaches zero.

Fig. 2(b) displays an opposite picture to the case discussed above. For Re=200, a small decrease and then a large increase in heat transfer is shown. The local Nusselt number is practically zero until it reaches the neighborhood of the discontinuity. Compared to Fig.



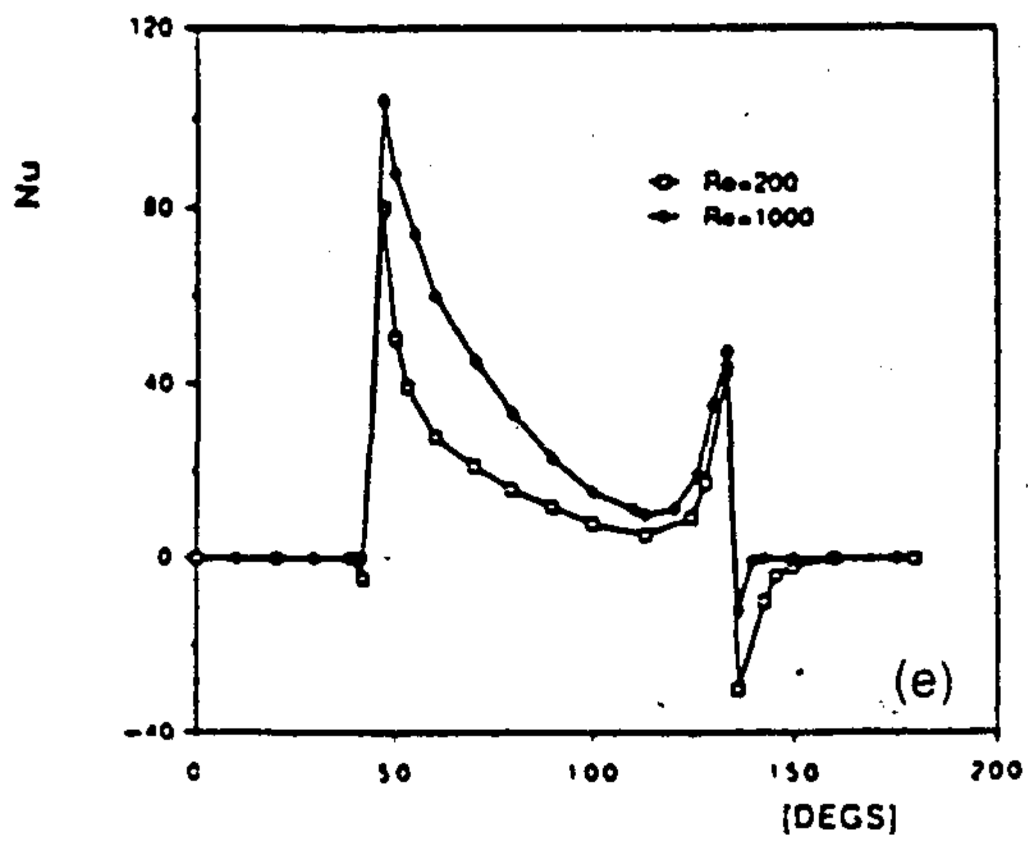


Fig.2 Effects of the step change in wall temperature on locat heat transfer in air, these cases compare to those shown in Fig.1

2(a), the portion that is heated at a constant temperature of $\Theta_w=2$ gives lower heat transfer for the most part except near the peak(from 90 to 100 degrees). This would definitely result in a lower average Nusselt number.

Figs. 2 (c) and (d) exhibit even larger increases and decreases at the point of discontinuity. This is due to a larger temperature change there. In Fig. 1, cases (c) and (d) have a step change in temperature that is twice as large as cases(a) and (b). Plot (e) corresponds to case (e) in Fig. 1. It shows all the characteristics at the point of discontinuity which are observed separately in each of the four cases discussed above.

The results for Re=1000 in each case show the augmentation in heat transfer. This is due to the convection dominance in heat transfer phenomena at higher Reynolds number. Diffusion becomes less important as expected. In Figs. 2 (a),(b),(c) and (e), there is no increase or decrease due to upstream diffusion. These should have been washed away by the convective fluxes.

Different nonuniform boundary conditions have produced different results, even though they all have the same mean wall temperature.

Some of the cases studied here have shown a significant change in the overall heat transfer in comparison to UWT or UHF bounday conditions. They also have revealed some dependency on the Reynolds number, which measures the relative importance of convection and diffusion fluxes in the heat transfer processes for a fluid stream.

Fig. 3 (a) shows the calculated average Nusselt numbers for both the UWT and UHF cases, as found in our earlier study. (As noted before the UHF case does not have the same average wall temperatures as the other cases). The experimental UHF results of Krall and Eckert[5] and the correlation of extensive data for the UWT case given by Morgan[8] are compared to our results. Extremely good agreement is shown. Fig. 3 (a) also compares UWT and UHF boundary conditions with the linear profiles in wall temperature. Differences between the curves grow as the Reynolds number is increased. The linear profile which locates the maximum temperature in the rear stagnation point gives the least heat transfer throughout. This is to be expected since the wake itself imposes a resistance to transfer phenomena. The UWT case lies between the upper and lower limits of heat transfer. A linear profile which locates the maximum at the front stagnation point shows the highest heat transfer of these cases in the range of Reynolds number covered here. However, no differences are detected between this linear profile and the UHF boundary condition up to Re=2000, Even at Re=3480, the differences are minute.

Step changes in wall temperature show profound effects on the average Nusselt number for the cases examined here. See Fig. 3 (b). The UWT boundary condition is compared in this figure with the cases of step change in wall temperature. Again differences grow percentagewise and in absolute magnitude as the

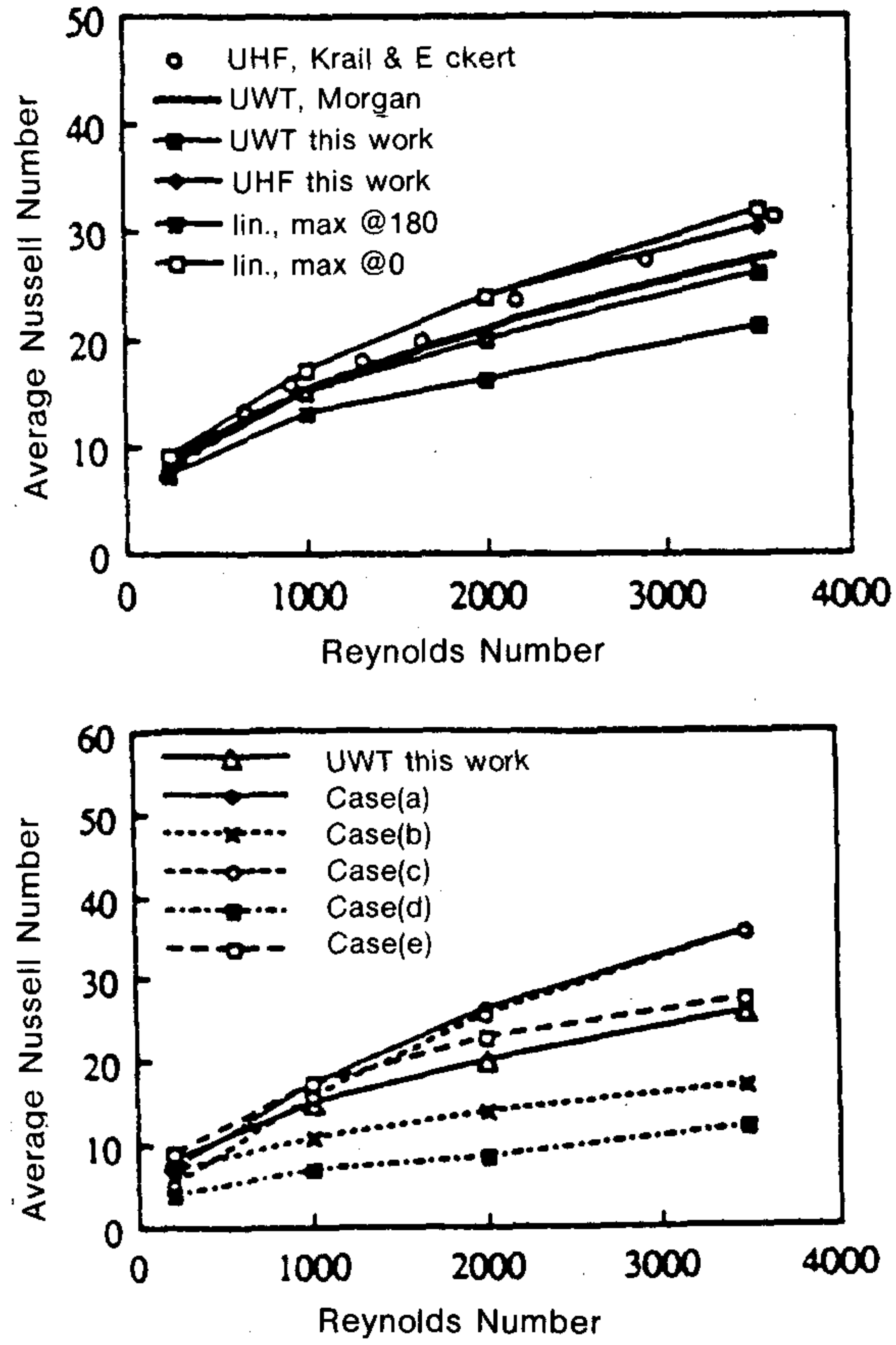


Fig.3 Average heat transter for (a) UWT and UHF data and calculations compared with the linear vartiation caiculations and (b) UWT caiculations compared to step cases shown in Fig.1; these resuits are for air

Reynolds number increases. For example, the case (b) in Fig. 1 shows 12.5 percent difference with the UWT case at Re=200. This becomes 34,6 percent at Re=3480. Beyond Re=1000, maximum heat transfer is attained when the front half is heated to maintain the dimensionless surface temperature at 2 (case (a) in Fig. 1). It also shows a rapid improvement in heat transfer for case (c) in Fig. 1. This reprents the case where the first quadrant is heated. At Re =3480, the value converges with that of the maximum case. Unless the rear half or quadrant are heated at a constant temperature, heat transfer is improved over the UWT case.

Although our analysis has ignored the unsteady nature of the flow on the rear portion of the cylinder, this does not render the solution insensitive to the boundary conditions in that region. This was shown in the earlier paper[3]. It can be seen by comparing the average heat transfer variation for cases where temperature changes occur on the downstream portion of the cylinder. See cases (b) and (d) in Figs. 1 and 3 (b).

4. CONCLUSIONS

A numerical study is presented of forced convection heat transfer over a very long circular cylinder in a uniform stream of air. Results are found using the stream function–vorticity approach. A series of situations were investigated where the surface temperature varied with angle around the cylinder. Included in this were linearly–varying temperatures and other situations where the surface temperature varied in a step–change manner. In all cases the variations were set so that the mean surface temperature remained same. This allowed easy comparisons between nonisothermal and isothermal cases for the same Reynolds number.

Some of the nonuniform surface temperature cases showed considerable differences in total heat transfer between one another. Differences between the various cases did not remain in the same relationships as the Reynolds number changed.

The practice used by some analysts when calculating nonisothermal heat transfer of invoking an average surface temperature with an isothermal correlation could be considerably in error. From the cases considered here the magnitude of this error could be on the order of 50%.

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ABSTRACTS SOLAR ENERGY VOL. 10, NO. 3, 1990

Thermally Stratified Hot Water Storage

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ABSTRACT

This paper deals with experimental research to increase thermal storage efficiency of hot water stored in an actual storage tank for solar application. The effect of increased energy input rate due to stratification has been discussed and illustrated through experimental data, which was taken by changing dynamic and geometric parameters. Ranges of the parameters were defined for flow rate, the ratio of diameter to height of the tank and inlet–exit water temperature difference. During the heat storage, when the flow was lower, the temperature difference was larger and the ratio of diameter to height of the tank was higher, the mementum exchange decreased. As for this experiment, when the flow rate was 8 liter/min, the temperature difference was 30℃ and the ratio of diameter to height of the tank was 3, the momentum exchange was minimized resulting in a good thermocline and a stable stratification. In the case of using inlet ports, if the modified Richardson number was less than 0.004, full mixing occured and so unstable stratification occured, which mean that this could not be recommended as storage through thermal stratification. Using a distributor was better than using inlet ports to form a sharp thermocline and to enhance the stratification. It was possible to get storage efficiency of 95% by using the distributor, which was higher than a storage efficiency of 85% obtained by using inlet ports in same operation condition. Furthermore, if the distributor was manufactured so that the mainpipe decreases indiameter toward the dead end to maintain constant static pressure, it might be predicted that further stable stratification and higher storage efficiency are obtainable(ie:more than 95%).

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ABSTRACT

When nonuniform thermal boundary conditions are imposed on the surface of a circular cylinder in crossflow, the heat transfer characteristics can be quite different compared to what is found for isothermal or constant heat flux boundary conditions. In the present analysis, two kinds of nonuniform boundary conditions along the circumference of the cylinder are considered in a uniform stream of air: step changes and linear profiles. Step changes in temperature can arise on the surface of an external, cylindrical, solar central receiver. As the working fluid(water) flows through the vertical tubes that ring the circumference of Solar One(a solar central receiver in Barstow, California), the solar flux on the receiver heats the water from a liquid to a superheated state. In this process, portions of the receiver panels, and thus portions of the circumference of the cylinder, function as a preheater, boiler, or superheater. Hence the surface temperature can vary significantly around the cylinder. Common engineering practice has been to use an average wall temperature with an isothermal cylinder heat transfer coefficient when estimating the convective loss in these kinds of situations.

Thermal Performance Analysis for the Low–Cost Solar System with Trickle–Collector

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ABSTRACT

Theoretical analysis for the thermal performance on the low–cost trickle collector, which is easy to manufacture and construct, has been performed. The results were in reasonably good agreement with those of the experiments. They have been applicable to predict long–term thermal performance on the low–cost solar collecting system. The dialogue type of computer program has been written based on the f–chart method and it can be used for designing a these collecting system, and investigating its economic feasibility.

The Estimation of Transpiration Rate of Crops in Hydroponic Culture in the Plastic Greenhouse

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Seoul National Univ. College of Agriculture Dept. of Agriculture Engineering.

ABSTRACT

The main objective of this study was to find the relationship between transpiration rate and en-